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An impeller for pumps

The invention relates to an impeller for pumps, in particular for radial pumps in accordance with the pre-characterising part of claim 1 and to a pump, in particular radial pumps with an impeller of this kind, and also to a method for operating a pump of this kind.

In centrifugal pumps with one or more impellers, the resultant of all the axial forces acting on the impeller or the impellers during operation can reach considerable values. Without additional measures this resultant force which is termed axial thrust, would be transferred via the pump shaft to the bearing and would correspondingly put a heavy load on this. Suitable constructional measures are known from the prior art for reducing the axial thrust, for example by means of a dual-flow pump arrangement with a mirror symmetrical design of the impellers, or in single flow pumps by providing sealing gaps on both sides of the impeller and openings in the impeller which connect the suction side with the reverse side of the impeller. A further possibility is to relieve the pump of axial pressure by means of a suitable relief device, such as a relief ring or piston, for example.

For the characterisation of a centrifugal pump impeller the specific rotational speed n_q is often used which is calculated in known manner from

the capacity or pump flow Q, head H and rotational speed n. In pumps for small specific rotational speeds n_q, of for example smaller than 15 min^{-1}, corresponding to a comparatively large head of greater than 50, 75 or 150 m and comparatively small capacities of smaller than 100, 50 or 25m^3/h for example, in particular in those kinds of process pumps, the problem arises that the above-mentioned measures for the axial thrust compensation can only be applied to a restricted extent. Thus a dual flow version of a pump for small capacities requires a considerable amount of additional cost and complexity in comparison with a single flow design. A balance piston is likewise comparatively costly and for this reason is mainly used in larger multi-stage pumps. For a single stage process pump, sealing gaps are normally provided on both sides of the impeller. An axial thrust compensation by means of sealing gaps on both sides of the impeller and openings in the impeller is, however, only possible when the impellers are closed. In the case of closed impellers with small specific rotational speeds n_q there is the problem of the manufacture, since the outlet widths of impellers of this kind are in the region of few mms and the manufacture of closed impellers with small outlet widths is difficult and expensive from the point of view of casting technology.

A further disadvantage of closed impellers is the high impeller friction losses (also called impeller side friction losses) and clearance gap losses which impellers of this kind have at small specific rotational speeds n_q. At a specific rotational speed n_q of 8 $min.^{-1}$ for example, the impeller friction loss alone amounts to 30% or more. Closed impellers for small specific rotational speeds n_q thus show a comparatively low degree of efficiency.

For the above-mentioned reasons, impellers for specific rotational speeds n_q smaller than 10 or 15 $min.^{-1}$ are often designed to be half-open. This has advantages from the point of view of casting technology and the wheel

friction of half-open impellers is considerably less than those of closed impellers. Half-open impellers for specific rotational speeds n_q less than 10 or 15 min.$^{-1}$ have the disadvantage however that the axial thrust compensation is difficult and the impeller friction losses are still very high.

An additional problem of closed and half-open impellers for small specific rotational speeds n_q is the tendency to instability in the part load region, i.e. impellers of this kind have a characteristic curve which is either unstable (corresponding to a falling characteristic curve, if the pump flow Q approaches 0) or only just stable (corresponding to a characteristic without a notable rise when Q approaches 0).

The object of the invention is to make available an impeller for pumps, in particular radial pumps, which facilitates a reliable axial thrust compensation even at rotational speeds n_q less than 10 or 15 min.$^{-1}$, which is comparatively economical to manufacture, which shows lower impeller friction losses when compared with a correspondingly dimensioned closed or half-open impeller and which has a stable characteristic curve in the part load region. A further object of the invention is to make available a pump, in particular a radial pump with an impeller of this kind and also a method for operating a pump of this kind.

This object is satisfied by the impeller defined in claim 1 and by the pump defined in claim 8 as well as by the method defined in claim 10.

The impeller for pumps in accordance with the invention, in particular radial pumps, includes one or more vanes and additionally an intermediate wall on which one or more vanes are provided on both sides respectively. At least one passage opening is formed in the intermediate wall in order to distribute a desired pump flow to the vanes on both sides of the

intermediate wall. The impeller preferably has a suction side which, in the inbuilt state of the impeller, is directed towards a suction opening of the pump and the vanes are preferably connected on the side of the intermediate wall directed away from the suction side to the suction side via at least one passage.

In a preferred embodiment the impeller has a hub and a plurality of passage openings in a region of the impeller adjacent to the hub. In a further preferred embodiment the impeller is open towards the suction side or towards both sides. The vanes preferably include shortened vanes, so-called splitter vanes. The impeller preferably has a specific rotational speed n_q in the region of 2 - 20 min.$^{-1}$, in particular in the region of 5 - 12 min.$^{-1}$.

The outlet edges of the vanes on the suction side and/or on the side of the intermediate wall remote from the suction side are chamfered. A chamfer of the vanes at the suction side which falls away towards the suction side is advantageous, while the vanes on the side of the intermediate wall remote from the suction side have outlet edges parallel to the axis. The chamfering of the vane outlet edges supports an ordered circulation which is particularly advantageous in the part load region.

The vanes on both sides of the intermediate wall are preferably so designed that, at part load, an ordered circulation occurs and the impeller has a characteristic curve which rises constantly, in particular rises constantly and clearly if the pump flow Q approaches 0. For this purpose the vanes are preferably designed differently on both sides of the intermediate wall, for example in that the vanes on both sides of the intermediate wall have different chamfers of the outlet edges and/or different vane outlet angles β_2 and/or different numbers of vanes.

The invention further includes a pump, in particular a radial pump with an impeller in accordance with the above description.

In the method according to the invention for the operation of a pump, in particular a radial pump with at least one impeller, the impeller additionally provided with an intermediate wall on which one or more vanes are provided on both sides and with at least one passage connecting the two sides. In the method according to the invention a desired pump flow is distributed over the vanes on both sides of the intermediate wall, preferably by a part of the pump flow being fed through the at least one passage opening from one side of the intermediate wall to the other side.

The impeller in accordance with the invention has the advantage that the division of the pump flow over the vanes on both sides of the intermediate wall makes a reliable axial thrust compensation possible. Furthermore, the impeller is distinguished by a stable characteristic curve and a stable part load behaviour. In the open form, the impeller in accordance with the invention facilitates a good rotor efficiency, since the wheel friction which accounts for a large part of the losses, in particular in impellers with low specific rotational speeds, is dispensed with. Thanks to the open form, vane output widths of a few mms are very easy to manufacture by means of casting and/or milling for example. The division of the pump flow over the vanes on both sides of the intermediate wall results in a reduced vane width which has a favourable effect on the casting properties and the natural frequency of the vanes. If needed the vane thickness can be reduced in comparison with a traditional impeller with the same pump head and the same capacity.

Further advantageous embodiments can be seen from the dependent claims and the drawings.

The invention will be explained in more detail with reference to the embodiments and with the help of the drawings, which show:

Fig. 1a a perspective view of an embodiment of an impeller in accordance with the present invention,

Fig. 1b the embodiment of an impeller shown in Fig. 1a from the front,

Fig. 2 a detailed view of a variant with a shortened intermediate wall, and

Fig. 3 an embodiment of a process pump with an impeller in accordance with the present invention.

Figures 1a and 1b show an embodiment of an impeller for pumps, in particular radial pumps in accordance with the present invention. The impeller 1 of the embodiment includes a hub 4, an intermediate wall 6, on which one or more vanes 2a, 2'a, 2b, 2'b are provided on both sides respectively as well as a passage 3 which, as shown in the Figures 1a and 1b, is formed in the shape of five passage openings in order to distribute a desired pump flow over the vanes 2a, 2'a, 2b, 2'b on both sides of the intermediate wall. In the embodiment the passage openings are formed in one region of the impeller adjacent to the hub 4. It goes without saying that other designs of the passage opening 3 are also possible. The impeller 1 expediently has a suction side 5a which, in the inbuilt condition of the impeller, is directed towards a suction opening of the pump and the vanes

2b, 2'b on the side of the intermediate wall remote from the suction side are connected with the suction side 5a via the passage opening 3, i.e. via the five passage openings in the example shown.

In the embodiment the impeller 1 is open towards both sides. It is however also possible that the impeller is only open on one side, the suction side 5a for example or is, if necessary, also closed. In the embodiment shortened vanes 2'a, 2'b, so-called splitter vanes are provided between the passage openings. In one embodiment the vane edges are chamfered at the outlet. In the variant shown in the Figures 1a and 1b, the vane outlet edges are double chamfered at the suction side 5a, firstly they have a chamfer towards the suction side 5a (the outlet edge is inclined to the axis) and additionally a profile (taper) on the pressure side. If necessary, as shown in Figures 1a and 1b, the vane inlet edges on the suction side 5a can be provided with a profile on the suction side. It is of course also possible to chamfer the vane outlet edges on the side 5b remote from the suction side or to provide a common chamfer for both sides jointly and also to provide blade outlet edges parallel to the axis on one or both sides.

In the embodiment all the vanes 2a, 2'a, 2b, 2'b extend straight ahead in the radial direction towards the outside. The vane inlet angle β_1 and the vane outlet angle β_2 thus amount to 90° in a first approximation. The actual vane outlet angle β_2 on the suction side 5a is slightly smaller than 90° as a result of the profile of the vane outlet edge on the pressure side. It is however also possible to provide curved vanes with a vane outlet angle β_2 smaller or greater than 90°. The vane inlet angle β_1 is advantageously smaller than the vane outlet angle β_2 and preferably smaller than or the equal to 90°. The vanes 2a, 2'a, 2b, 2'b advantageously have a vane inlet width b_1, at least at one side, preferably at the suction side 5a, which is larger than the vane outlet width b_2. The vane outlet widths b_1 on the

suction side 5a and the side 5b remote from the suction side are expediently designed in such a way that a good suction capability is obtained. The ratio of the vane widths on the suction side 5a and on the side 5b remote from the suction side can be selected within wide limits. In a typical embodiment the vane widths on both sides are approximately the same size.

In a preferred embodiment the impeller 1 has a specific rotational speed n_q in the range of 2 - 20 min.$^{-1}$, preferably in the range of 7 - 12 min^{-1}.

In the embodiment shown in Fig. 2, the impeller includes an intermediate wall 6, the outer diameter of which is smaller than the outer diameter of the impeller. On each side of the intermediate wall 6 there are respective vanes 2a, 2b which join at the outer end so that the two vanes have a common outlet edge 7. The intermediate wall 6 and the vanes 2a, 2b are arranged on a hub 4. A passage opening 3 is additionally formed in a region of the impeller adjacent to the hub 4 which connects the two sides of the intermediate wall 6 in a fluid conducting manner. Since the intermediate wall 6 in this embodiment only extends over a part of the vane length, the intermediate wall can also be termed an intermediate bridge if desired.

Fig. 3 shows an embodiment of a pump, preferably a process pump with a rotor in accordance with the present invention. The pump 10 of the present embodiment includes an impeller 1, for example an open impeller in accordance with the above-described embodiment, with a hub 4, an intermediate wall 6 on which vanes 2a, 2b are provided on both sides and with a passage opening 3, in order to distribute a desired pump flow over the vanes 2a, 2b on both sides of the intermediate wall 6. The pump 10 further includes a housing 11, an inlet or suction opening 12, a ring

channel 17a which adjoins the outside of the impeller 1, a diffuser insert 17 and an annular cavity17b which opens out into an outlet or pressure connection 13. By means of a modified insert 17, a spiral or an annular cavity can be produced which is directly connected to the outlet or pressure connection or stub 13. By adapting the diffuser insert it is possible to compensate for the generally considerable radial thrust in pumps with low specific rotary speeds. A cast housing without an exchangeable insert can of course also be used as a guide mechanism.

The pump further includes a shaft 14, a shaft seal 15, for example a stuffing box and bearings 16a, 16b for supporting the shaft. In the embodiment the bearings 16a, 16b are designed as ball bearings which apart from radial forces, can also take up axial forces, depending on the design. In Fig. 3 the bearings 16b on the right-hand side of the drawing are specially constructed to take up radial and axial forces for example, so that any eventually present, not completely compensated residual component of the axial thrust can be taken up without any problem.

In a preferred embodiment of the method in accordance with the invention for the operation of a pump, for example a pump in accordance with the embodiment described above, with at least one impeller, the impeller is additionally provided with an intermediate wall 6 at which one or more vanes 2a, 2b are provided on both sides and with a passage opening 3 which connects the two sides of the intermediate wall 6 in a fluid conducting manner. In this embodiment of the method in accordance with the invention a desired pump flow is distributed over the vanes 2a, 2b on both sides of the intermediate wall 6, in that a part of the pump flow, 5% to 75% for example and preferably approximately 50% is fed from one side of the intermediate wall 6 to the other side. The pump flow of the impeller 1 is advantageously distributed onto the vanes 2a, 2b on both sides in such

a way that the axial thrust is compensated. The above-described embodiment of the method in accordance with the invention can be used in one-stage pumps and also in multi-stage pumps.

The impeller in accordance with the invention has the advantage that the distribution of the pump flow on the vanes on both sides of the intermediate wall facilitates a simple, and practically complete axial thrust compensation. The above-described impeller is distinguished further by a stable characteristic curve and good efficiency. The impeller in accordance with the invention can be used in one-stage pumps and also in multi-stage pumps.